THERMAL POWER PLANTS

PROSPECTS FOR INCREASING THE EFFICIENCY OF GAS TURBINE AND STEAM-GAS UNITS[1]

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This is a parametric study of the performance of the new generation of prospective gas turbine units and steam-gas units based on them over a wide range of variation in the parameters that determine the specific work and efficiency of a steam-gas unit. Optimum values are obtained for these parameters and the tasks needed for future development of Russian gas turbine and steam-gas units capable of competing with the best foreign units are formulated.

Keywords: gas turbine unit; steam-gas unit; efficiency; compression ratio; compressor; turbine; cooling; performance

Historically an enormous number of gas turbine cycles have been studied. Here the major emphasis has been on intermediate cooling of air during compression and heating of combustion products during expansion (up to four or five stages of compression and expansion), regeneration of heat (direct and chemical, with conversion of methane into H_2 and CO), and various forms of introducing water into the working medium of gas turbine units.

These studies are no longer of great interest.

Most of the gas turbine units in practical use have been designed with a simple thermodynamic cycle: one stage of compression to a pressure of 1.5 – 3.0 MPa, input of heat by combustion of the fuel, and a single stage of expansion.

Gas turbines with a simple cycle are fully developed and a we have a large amount of experience in their design, production, and operation. A conventional design for high power gas turbine units has evolved.

Any innovations beyond this experience require a long time and involve great risks, so it is best that prospective high power gas turbines be developed for a simple thermodynamic cycle.

The thermal performance of a gas turbine unit is affected to the greatest extent by the initial temperature of the gases entering the turbine to perform work. In other countries commercial gas turbines are manufactured with initial temperatures of 1350 – 1500°C, and gas turbines with higher temperature levels of 1600°C have been designed and are in use. A gas turbine for 1700°C is under development.

In order for Russian gas turbine units to be competitive at the level of 2020, they will also have to be designed for initial gas temperatures $t_{1t} \geq 1600$°C. Many problems have to solved in order to achieve these higher temperatures:

— develop effective systems for cooling of components;

— improve the materials and coatings for them;

— develop and explore economically acceptable technologies for manufacture of components, especially turbine blades; and,

— create fuel ignition systems with acceptable emissions of NO_x and CO during operation.

Another important parameter is the pressure increase in the cycle. For given values of the initial gas temperature and internal performance indicators affecting the perfection of the gas turbine unit (the efficiency of turbine machinery, cooling air flow rates, pressure drops in loops, etc.), there are two optima for the degree of compression π_c — with respect to specific work and with respect to efficiency.

The optimum with respect to specific work, for which the specific cost of the system is minimal, is 20 – 25 in prospective gas turbine units. It is also close to the optimum with respect to efficiency of binary steam-gas units with the corresponding gas turbine units. The specific work has a smooth dependence on the degree of compression near the

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optimum and the optimum value can be adjusted taking the other important factors into account.

In particular, a temperature t2t of the exhaust gases from the turbine at a level of 630 – 650°C is desirable as it permits a free choice of the fresh steam parameters in the steam loop (e.g., preheating to 600°C) with a reduction in the heating surface in the steam superheaters.

The optimum degrees of compression with respect to the efficiency of a gas turbine unit are considerably higher and are not realized in power generating gas turbines.

The first advances in gas turbine engines 80 years ago began with water cooling of the turbine blades. In the USSR work in this area with large experimental units was undertaken in 1965 – 1975 at VTI, TsKTI, and MVTU. Somewhat later a gas turbine unit with water cooling of the turbine was designed by General Electric (US) based on comprehensive studies and engineering development. As opposed to the Russian studies, in which water was supplied and removed in the liquid state, the General Electric design was based on evaporation in the cooling channels and steam was injected into the flow-through section, where it was mixed with combustion products.

The design with water cooling of the turbines was never realized in practice.

Programs to create high power gas turbine units with steam cooling of the turbines were undertaken in the 1990's in the US and Japan.

General Electric built the 9H gas turbine and Mitsubishi, the 501H. These gas turbines were tested, and a steam-gas unit with the 9H turbine has been run for a long time at an electric power plant. Closed steam cooling of the gas collectors in the combustion chamber and nozzle apparatus of the first stages of the turbine is used in Mitsubishi's 501G and 701G gas turbines.

The experience gained from the use of steam cooling has shown that it makes the design and construction more complicated and limits the maneuverability of the gas turbine unit. The two companies that have produced these gas turbines have modifications of similar gas turbine units with cooling entirely by air that are basically available at present.

Given the results of work on cooling the components of gas turbines with water and the industrial mastery of steam cooling, it developing prospective gas turbine units it is appropriate to concentrate on air cooling of the components.

The most important question is then the efficient consumption of cooling air. To solve this problem it is necessary to develop the most efficient:

— designs for internal convective loops for blade cooling,

— systems for release of air on surfaces around which gas flows in order to create protective films (profiling the output apertures, placing them in accordance with the actual gas flow conditions), and

— designs for air feed loops (intake of air at the lowest pressure, active control of it by switching the intake locations depending on the operating mode for the gas turbine unit, sequential use of air first in nozzle and then in working blades, swirl at the input of rotating parts, densifications, etc.).

It is necessary to estimate the efficiencies of the different methods for reducing the temperature of the cooling air and the prospects for the use of porous materials and composite designs.

The possibility of using steam for cooling individual parts of the stator is not excluded.

The blades of aviation gas turbine engines work at temperatures up to 1000°C and fixed power generating gas turbine units, up to 850 – 900°C. The high temperature components of modern gas turbine units are made of refractory complex alloys based on nickel. Similar alloys, with improvements for better operating and engineering properties should also be used in the prospective high power gas turbine units. Blade surfaces in contact with gas must be coated to protect them from oxidation and corrosion. Another purpose for external coatings is to improve the thermal resistance of the blade wall during heat transfer between the gases and external air and thereby to reduce the temperature of the stress bearing metal. Ultimately, a complicated multilayer coating is applied to the blade; it includes a ceramic layer that must also tolerate thermal cycling and the impact of impurity particles. The cooled turbine blades with complex internal loops for movement of cooling air, complicated external shapes, shafts, and hoop racks are fabricated by precision casting techniques: equiaxial for ordinary conditions, and with directed crystallization for more severe conditions and single crystals for extreme conditions.

Building a domestic, high power gas turbine unit will require mastery of the techniques for single crystal casting of the blades for the first and second stages of the turbine.

The design and fabrication of the working blades and disk of the last stage of the turbine may require correction of the durability standards currently in force in Russia, which must computationally and experimentally validated.

The technologies for depositing and monitoring the suitability of multilayer thermal-barrier coatings are also undeveloped in this country. Developing and mastering these technologies are also needed to ensure a long service life of the blades through recovery of their properties and repeated coating.

Of greatest importance for evaluating the different ways of enhancing the economic performance of gas turbine units, even when these ways have been demonstrated in construction and confirmed in practice, is the existence of economical technologies for doing these things.

The development of a high power gas turbine unit will certainly require a three dimensional design for the flow through part of the compressor and turbine, as well as accounting for end effects and mixing with cooling air.

The design of blades, interblade channels, and meridional surfaces must be optimized taking the aerodynamics into account, as well as cooling of the corresponding components, especially if this involves the use of protective air films. Leaks through radial gaps during operation must be

minimized with the aid of special systems for regulating the gaps that ensure their conservation during cycling of the gas turbine.

When designing the compressor it is necessary to balance the head and economy of the stages so that with a small number of them (and a moderate cost for the compressor) it is possible to a maintain high efficiency and reserve of stability at the nominal rotation speed.

With steerable director apparatus it is desirable to reduce the compressor capacity by 40% or more with acceptable compressor efficiencies.

The characteristics of the stages and compressor sections should ensure rotation of the gas turbine unit during startup by an external source whose power does not exceed 2% of the nominal gas turbine power.

The prospective high power gas turbine units must ensure complete combustion of various gaseous fuels (natural, byproduct, liquefied, and synthetic gases) with production of no more than 50 mg/m^3 of nitrogen oxides under normal conditions. Designs and studies of the ignition systems should be carried out with a view to further reductions in the NO_x emissions to 15 – 20 mg/m^3 under normal conditions. To sustain complete combustion and the specified NO_x emissions over a wide range of loads and external conditions, the fuel should be distributed over several channels with discrete shutting on or off of each depending on the operating mode.

It is also appropriate to develop a system for burner bypass of part of the air when the load is reduced with possible stepwise combustion and air ballast by combustion products. A low emission combustion scheme for burning of liquid fuel or vapor with large air excess should also be developed.

At intermediate gas temperatures above 1600°C the yield of thermal NO_x begins to rise rapidly. The difficult problem of maintaining the attained level of nitrogen oxides without air ballast arises, for example, by recirculation of exhaust gases or the introduction of steam, the use of a catalytic combustion stage, or catalytic nitrogen removal.

This parametric study is carried out for a single-shaft gas turbine with a simple cycle at maximum power operating with a constant rotation frequency of 3000 rpm. The calculations are done per unit of air feed through the compressor (1 kg/sec). The air and combustion products are considered to be ideal gases whose thermodynamic properties depend only on temperature. These dependences are taken into account by the method of ref. 1. The molecular weight and composition parameter in that method are determined at each point from the temperature rise of the medium in the combustion chamber, which depends uniquely on the fuel/air ratio. These parameters are constant for air.

Based on an analysis of the technical level of current commercial and developmental high power generating gas turbines [2] we have carried out these parametric studies with

— gas temperatures at the turbine inlet, t_{1t}, ranging from 1300 – 1700°C with a step size of 100°C, and

— compression ratios π_c ranging from 15 – 30 with a step size of 3.

Since the gases consumed in the turbine are used to produce steam which then expands in the steam turbine, the range of variation in their temperatures is limited to 560 – 660°C, which ensures a yield of steam with temperatures of 520 – 620°C and does not simultaneously create new problems with materials for the working blades in the last turbine stage.

Advances by domestic and foreign companies and organizations working on turbine machinery for gas turbine units, the results of tests at VTI on real gas turbine units, and preliminary calculations were taken into account when choosing values for the internal characteristics of the components of the gas turbine unit. Since the blades are quite high for high unit powers, the calculations include values of the internal efficiencies of the turbines and compressor that are close to the maximum attainable.

The polytropic compressor efficiencies are taken to be 90.0 – 93.5%. For compression ratios of 15 and 30, these correspond to adiabatic efficiencies of 84.5 – 85.7 and 88.3 – 89.3%, respectively.

The efficiencies of 90 – 96% assumed for the turbines consider only the aerodynamic perfection of the flow through portion and can be compared with the efficiencies of the medium pressure cylinders of steam turbines.

The air feed rate for cooling $\overline{G}_{\text{cool}}$ varies by 10 to 25%. To estimate its effect on the efficiency of the gas turbine unit it is assumed nominally that:

— all the cooling air flow is collected after the compressor;

— half of the flow is expended in cooling the nozzle blades in the first stage; this air is mixed with the main combustion product flow in the axial gap between the nozzle and working blades, reduces their temperature, and then expands together with them with the specified turbine efficiency; and,

— the other half of the cooling air does not operate in the turbine; this air is mixed with the main flow after the turbine and reduces the temperature of the exhaust gases from the turbine.

Under these conditions the results can be brought into agreement with other computational and experimental data.

The pressure losses in the gas turbine loops are assumed equal to 8%.

Since the main purpose of the gas turbine unit is to be used as a part of prospective steam-gas installations, calculations of two- and three-loop steam sections have been carried out and their efficiencies have been determined as a function of the live steam temperature depending on these efficiencies.

In essence, it was more convenient to use data on the specific work of the steam turbines in steam-gas units from leading international companies. The results of an analysis of the catalog data [3] are listed in Table 1.

All the systems included in Table 1 are built with a three pressure steam loop and intermediate steam superheat. The

specific work of the steam turbine was defined as the difference between the net power of the steam-gas unit and the power of one or two autonomously operating gas turbine units, i.e., taking into account all the in-house needs and the enhanced pressure loss after the gas turbine owing to the presence of a boiler-utilizer.

The analytic expression $N_{st} = 6 \times 10^{-3} t_{2t} - 0.0141$ for calculating the specific work of a steam turbine (in MJ/kg) as a function of the temperature of the effluent gases from the gas turbine unit was derived from the catalog data.

The gas turbine unit cycle was calculated for different combinations of the defining parameters:

— gas temperature t_{1t} at the outlet from the combustion chamber;

— compression ratio π_c in the compressor;

— cooling air flow rate $\overline{G}_{cool}$;

— turbine efficiency η_t; and,

— compressor efficiency η_c.

At the start of the calculation these parameters are chosen to be the minima within their assigned range of variation.

The results of the parametric study for initial gas temperatures ahead of the turbine (at the outlet of the combustion chamber) of 1500, 1600, and 1700°C were represented by analytic fits for the temperature of the effluent gases from the gas turbine unit (GTU), the specific work of the gas turbine unit, and the efficiencies of the gas turbine unit and steam-gas unit (SGU):

$$\frac{\eta_{GTU}}{100} = a_0 + a_1 \frac{t_{1t} + 273}{1973} + a_2 \left(\frac{t_{1t} + 273}{1973} \right)^2 +$$

$$+ a_3 \frac{\pi_c}{27} + a_4 \left(\frac{\pi_c}{27} \right)^2 + a_5 \eta_t + a_6 (\eta_t)^2 +$$

$$+ a_7 \overline{G}_{cool} + a_8 (\overline{G}_{cool})^2 + a_9 \eta_c;$$

$$N_{GTU} = a_0 + a_1 \frac{t_{1t} + 273}{1973} + a_2 \left(\frac{t_{1t} + 273}{1973} \right)^2 +$$

$$+ a_3 \frac{\pi_c}{27} + a_4 \left(\frac{\pi_c}{27} \right)^2 + a_5 \eta_t + a_6 (\eta_t)^2 +$$

$$+ a_7 \overline{G}_{cool} + a_8 (\overline{G}_{cool})^2 + a_9 \eta_c;$$

$$\frac{\eta_{SGU}}{100} = a_0 + a_1 \frac{t_{1t} + 273}{1973} + a_2 \left(\frac{t_{1t} + 273}{1973} \right)^2 +$$

$$+ a_3 \frac{\pi_c}{27} + a_4 \left(\frac{\pi_c}{27} \right)^2 + a_5 \eta_t + a_6 (\eta_t)^2 +$$

$$+ a_7 \overline{G}_{cool} + a_8 (\overline{G}_{cool})^2 + a_9 \eta_c;$$

$$\frac{t_{2t} + 273}{970} = a_0 + a_1 \frac{t_{1t} + 273}{1973} + a_2 \left(\frac{t_{1t} + 273}{1973} \right)^2 +$$

$$+ a_3 \frac{\pi_c}{27} + a_4 \left(\frac{\pi_c}{27} \right)^2 + a_5 \eta_t + a_6 (\eta_t)^2 +$$

$$+ a_7 \overline{G}_{cool} + a_8 (\overline{G}_{cool})^2 + a_9 \eta_c.$$

The coefficients for these fits are listed in Table 2.

The results of the parametric study for a gas temperature of 1600°C at the inlet to the turbine with a cooling air feed of 20 or 25% of the feed at the inlet to the compressor are shown in Figs. 1 – 3.

TABLE 1. Calculating the Specific Work of Steam Turbine Units as Parts of Steam-Gas Units

Parameter	Type of gas turbine unit and manufacturer							
	GT-26 Alstom	V94.3A Ansaldo	9FA General Electric	9FB General Electric	701F4 Mitsubishi	701G2 Mitsubishi	SGT5-4000F Siemens	SGT5-8000H 4000F Siemens
Net steam-gas unit power, MW	424/850.33333	411/820.3	390.8/786.9	437.2/872	464.5/932.1	498/999.4	423/848	570/1140
Net steam-gas unit efficiency, %	58.3/58.5	57.8/57.6	56.7/57.1	58.6/58.4	59.5/59.7	58.4/58.6	58.4/58.5	60.0/60.0
Gas-turbine unit power, MW	292.1/584.2	285/570	256.2/512.3	284.2/568.4	312.1/624.2	334/668	288/576	375/750
Steam turbine power, MW	131.9/267.9	126.6/250.3	134.6/274.3	153/303.6	152.4/307.9	164/331.4	135/272	195/390
Gas temperature after gas turbine, °C	615	572	599	614.7	597	587	577	625
Gas feed rate after gas turbine, kg/sec	653.8/130.7.6	690.5/1381.0	643.5/1287.0	655.7/1311.4	703.2/1406.4	737.8/1475.6	692.8/1385.6	830/1660
Specific work of steam turbine, kJ/kg	201.8/204.9	183.3/181.2	209/213	233/235	216/219	222.3/224.6	189/191	235/235

Note. The numerator is for one gas turbine unit in a steam-gas unit and the denominator, for two.

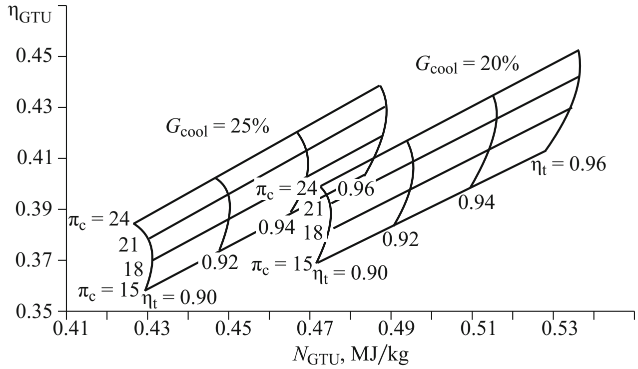

Fig. 1. Efficiency of the gas turbine units as a function of their specific work for t_{1t} = 1600°C.

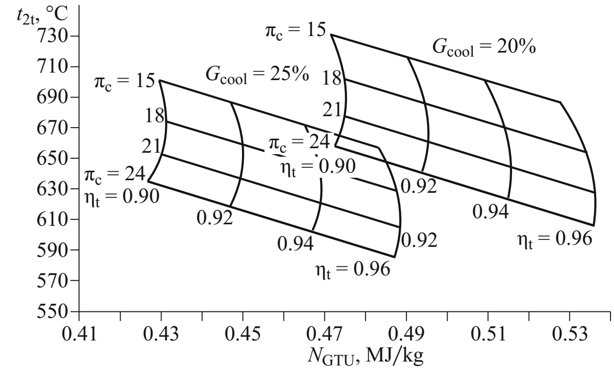

Fig. 2. Efficiency of the gas temperature after the turbine as a function of their specific work for t_{1t} = 1600°C.

Gas turbine units with a temperature of 1500°C have already been constructed and are in use. At 1700°C there are still some unclarified fundamental questions (the balance of air cooling the turbine for traditional cooling methods, NO_x emissions). A temperature level of around 1600°C is of interest for the next decade and is attainable with technologies that have been developed for a long time and employ ideas from aircraft engine construction. Thus, we have examined the results of a parametric study of gas turbine units at this temperature level in more detail.

In the realistic range of 92 – 94% turbine efficiencies and relative cooling air flows of 20 – 25% with an initial gas temperature of 1600°C it is possible to reach a gas turbine unit efficiency of 39 – 42% with a specific work of 450 – 520 kJ/kg (the power of the gas turbine unit with an air feed of 900 kg/sec is 400 – 470 MW). The lower values correspond to a moderate turbine efficiency with a large cooling air flow rate and the upper values, to a higher efficiency and more efficient cooling (see Fig. 1).

The optimum compression ratios for the air in the compressor with respect to specific work are 18 – 21.

Increasing the turbine efficiency by 1% increases the specific work by 2.2% and reduces the specific heat expenditure by 2.1%.

Reducing the relative feed rate of cooling air by 1% in the range from 25 – 20% increases the specific work by the same 2.2%, and lowers the specific heat expenditure by 0.65%.

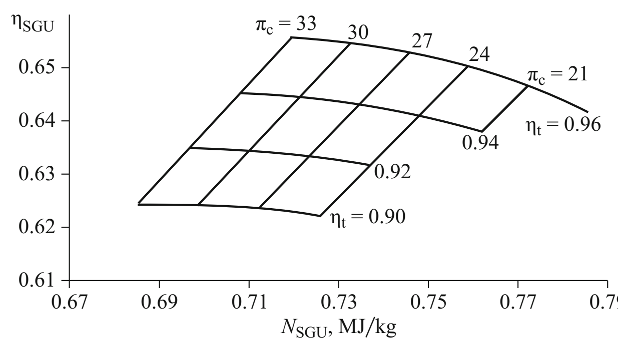

Fig. 3. Efficiency of the steam-gas units as a function of their specific work for t_{1t} = 1600°C and $\overline{G}_{cool}$ = 20%.

The efficiency of the turbine and the relative cooling air feed rate are the major internal factors governing the performance (thermal efficiency) of a gas turbine unit.

In order to maintain moderately high, at a level of 630 – 650°C, gas temperatures after the gas turbine, it is appropriate to settle on a compression ratio of 20 – 21 for a relative cooling air feed rate $\overline{G}_{cool}$ = 25% and 22 – 24 for $\overline{G}_{cool}$ = 20% (Fig. 2).

TABLE 2. Coefficients in the Approximate Fits for the Calculated Dependences of the Parameters of Gas Turbine Units and Steam-Gas Units

Coefficient	η_{GTU}	N_{GTU}	η_{SGU}	t_{2t}
a_0	–0.17068488	–0.32514193	–0.030132941	0.0074516459
a_1	–0.32938657	–0.90613608	0.26449707	1.4450002
a_2	0.28110384	1.0625579	0.011433621	–0.26178962
a_3	0.15140178	0.0064551261	0.045231134	–0.33253712
a_4	–0.055555579	–0.01967456	–0.018853401	0.096383808
a_5	–0.59139022	–0.65800058	–0.21597016	1.2827397
a_6	0.81212566	0.89115578	0.37039355	–1.1536563
a_7	–0.36848944	–0.94882804	–0.20380912	–0.12222551
a_8	0.10573756	0.084142958	–0.17047378	–0.6947295
a_9	0.52148368	0.82791688	0.33267982	–0.14502358

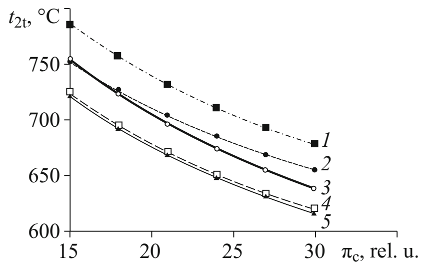

Fig. 4. Effluent gas temperature as a function of compression ratio for $t_{1t} = 1700$°C: *1*, $\overline{G}_{cool} = 20\%$, $\eta_t = 0.90$; *2*, $\overline{G}_{cool} = 25\%$, $\eta_t = 0.90$; *3*, $\overline{G}_{cool} = 25\%$, $\eta_t = 0.92$; *4*, $\overline{G}_{cool} = 20\%$, $\eta_t = 0.94$; *5*, $\overline{G}_{cool} = 25\%$, $\eta_t = 0.92$.

For the gas turbine unit performance examined here, steam-gas unit efficiencies ranging from 61.5 (for $\eta_t = 0.92$ and $\overline{G}_{cool} = 25\%$) to 64% (for $\eta_t = 0.92$ and $\overline{G}_{cool} = 20\%$) are attainable. The efficiency of the steam-gas unit depends weakly on the compression ratio (Fig. 3). As it is raised the efficiency increases slightly but the specific work falls off significantly more.

As for the gas turbine unit, the main factors influencing the efficiency of the steam-gas unit are the efficiency of the gas turbine and the flow rate of air for cooling it. The efficiency of the compressor has less influence although, of course, it should also be taken into account.

For high temperatures (about 650°C) of the gases at the inlet to the boiler-utilizer, high pressure(14 MPa) or even supercritical (24 MPa) steam can be generated with main and intermediate superheating to 600 – 620°C. The specific work (per 1 kg/sec of gases entering the boiler-utilizer) of the steam turbine for these parameters is about 240 kJ/kg, and the specific work of the steam-gas power generating unit will be 700 – 750 kJ/kg.

When the temperature of the initial gases is raised to 1700°C it is necessary to increase the compression ratio in the compressor in order to maintain acceptable temperatures for the gases consumed in the turbine.

The temperature of the effluent gases is plotted in Fig. 4 as a function of the expansion (compression) ratio for a constant (1700°C) gas temperature at the inlet to the turbine and different efficiencies of the turbine machinery and cooling air intake.

As the compression ratio in the compressor is increased and, thereby, the expansion in the turbine, the gas temperature at the outlet is reduced, so that for the chosen limitations the compression ratio must equal 27 – 30. Increasing the cooling intake and efficiency of the turbine will reduce the temperature of the gases used up in it. Thus, the efficiency of the gas turbine unit when 25% of the air is extracted for cooling can reach 41.0 – 43.5% and the specific work of the gas turbine unit, 495 – 520 kJ/kg.

Net steam-gas unit efficiencies of 63.5 – 65.0% can be attained with these types of gas turbines.

CONCLUSIONS

1. A model, algorithm, and program have been developed for finding the specific work and efficiency of gas turbine units as functions of the controlling parameters for the gas turbines: gas temperature at the turbine inlet, compression ratio in the compressor, cooling air flow rate, and turbine machinery efficiencies.

2. An initial gas temperature around1600°C should be chosen for developing the next generation of gas turbines. These temperatures can be achieved by making use of the latest advances in basic and applied science and technological practice.

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